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IN-SITU CHARGE DETERMINATION FOR VAPOR CYCLE SYSTEMS IN AIRCRAFT (POSTPRINT)

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14. ABSTRACT
The Air Force Research Laboratory (AFRL), in cooperation with the University of Dayton Research Institute (UDRI) and Fairchild Controls Corporation, is operating an in-house advanced vapor compression refrigeration cycle system (VCS) test rig known as ToTEMS (Two-Phase Thermal Energy Management System). This test rig is dedicated to the study and development of VCS control and operation in support of the Energy Optimized Aircraft (EOA) initiative and the Integrated Vehicle ENergy Technology (INVENT) program. Previous papers on ToTEMS have discussed the hardware setup and some of the preliminary data collected from the system, as well as the first steps towards developing an optimum-seeking control scheme. A key goal of the ToTEMS program is to reduce the risk associated with operating VCS in the dynamic aircraft environment. One of the key questions regarding the operability of VCS in aircraft which will be addressed is the in-situ measurement of refrigerant charge within the VCS system. Several potential methods of determining whether an appropriate charge of refrigerant exists will be discussed. An appropriate charge level is one which enables safe and efficient operation of the VCS over its designed operating envelop.

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VCS, vapor cycle system, VCS control, refrigeration, refrigerant charge level, in-situ refrigerant level monitoring

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ABSTRACT

The Air Force Research Laboratory (AFRL), in cooperation with the University of Dayton Research Institute (UDRI) and Fairchild Controls Corporation, is operating an in-house advanced vapor compression refrigeration cycle system (VCS) test rig known as ToTEMS (Two-Phase Thermal Energy Management System). This test rig is dedicated to the study and development of VCS control and operation in support of the Energy Optimized Aircraft (EOA) initiative and the Integrated Vehicle ENergy Technology (INVENT) program. Previous papers on ToTEMS have discussed the hardware setup and some of the preliminary data collected from the system, as well as the first steps towards developing an optimum-seeking control scheme. A key goal of the ToTEMS program is to reduce the risk associated with operating VCS in the dynamic aircraft environment. One of the key questions regarding the operability of VCS in aircraft which will be addressed is the in-situ measurement of refrigerant charge within the VCS system. Several potential methods of determining whether an appropriate charge of refrigerant exists will be discussed. An appropriate charge level is one which enables safe and efficient operation of the VCS over its designed operating envelope. The implications of these charge states, as applied to both static and dynamic VCS, will be addressed. To be effective for determining whether appropriate charge exists in an aircraft VCS, any potential method will need to operate in real-time and utilize in-situ sensors with current and recorded data, as opposed to relying on external gauges or specialty instrumentation that might be connected during scheduled maintenance. The method chosen may also indicate the minimum sensor suite needed for an aircraft VCS. Preliminary data will be discussed to illustrate the effect of charge level on VCS performance.

INTRODUCTION

More waste heat will be generated in next generation aircraft for several reasons. These include, for example, the increased use of electric architectures, more electronics, and new weapons. The thermal challenge will be further aggravated by a simultaneous reduction in heat sink due to greater use of composite materials, reduced fuel flow due to more efficient engines, and any fuel cooling limitations. A change in acceptable aircraft thermal management technologies must be adopted to meet growing thermal challenges. Air compression systems (ACS) have long been used to provide cabin air pressurization and cooling, due to an abundant supply of high pressure air from the gas turbine engine. With the recent trend toward the use of bleedless engine architectures, aircraft designers are exploring alternatives, such as an electrically driven ACS and vapor compression system (VCS). In general, an ACS has a lower efficiency than a VCS but can reject heat at a higher temperature [1]. This is due to the working fluid in an ACS neither acquiring nor rejecting heat at constant temperature as in the case of a VCS.

The phase change processes within a VCS provides significantly higher heat transfer rates than those available in gas-to-gas heat exchangers of similar volume and weight. Thus, a VCS which would be used in an aircraft thermal management system would tend to be smaller in volume and weight than an ACS.

A VCS used in a sixth generation aircraft thermal management system will have characteristics that are strikingly different from a traditional stationary VCS found in a home refrigerator, or building air conditioning system. These systems undergo relatively slow transients (on the order of 10's of minutes to hours) of relatively small amplitude. For example, the transient loads associated with a home air conditioner result from changes in external air temperatures which are a consequence of gradual weather changes and diurnal cycling. However the incorporation of a VCS in an adaptable, energy efficient thermal management system of an aircraft will require a VCS that can respond to rapid load transients (on the order of seconds) which may likely have large amplitudes. A VCS used as part of an aircraft thermal management system will have more stringent weight and volume restrictions relative to traditional stationary VCS. In addition, the cooling sink for an aircraft may change over a large temperature range, and its availability will depend on whether fuel or ambient air is used.

Refrigerant systems have the potential to gradually leak over long periods, and the refrigerant charge may eventually become inadequate. Symptoms of inadequate charge for large stationary chillers include a reduction in cooling capacity due to reduced availability of liquid for evaporation. In addition, the coefficient of performance (COP) may likely decrease since more compressor work is required to provide the same cooling load as with a fully charged system. On the other hand, a VCS system may be overcharged. Overcharging may lead to complete filling of the condenser with liquid which reduces the surface area for condensation and may potentially lead to excessively high pressures and, ultimately, compressor damage. There are well established design rules to ensure an acceptable charge for stationary chillers and refrigeration systems [2]. Kim and Braun [3] have demonstrated a charge detection and sensing system for steady-state operation. To the authors' knowledge, there have been few (if any?) studies to define an acceptable charge for a VCS in which significant load transients occur on the order of seconds. An acceptable charge might not result in an optimum COP under all conditions. The best charge will optimize the COP for most conditions and still maintain operation through a sharp transient. In addition, in situ sensing of the charge is necessary as the use of traditional detection methods such as viewing through a sight glass or occasional weighing of the refrigerant are impractical for use on an aircraft.

The purpose of this paper is to address the issues of appropriate charge in an aircraft VCS. An appropriate charge level is one which enables safe and efficient operation of the VCS over its designed operating envelope. One key issue regarding the operability of VCS in aircraft which will be addressed is the in-situ measurement of this refrigerant charge within the VCS system. To provide real time information, any potential method will need to use in-situ sensors with current and recorded data, as opposed to relying on instrumentation that might be connected during scheduled maintenance. The method chosen may also indicate the minimum sensor suite needed for an aircraft VCS. Several different possible refrigerant charge states may be identified, including over-charged, correctly-charged, slightly under-charged, and under-charged. The implications of these charge states, as applied to both static and dynamic VCS, need to be addressed. The Air Force Research Laboratory (AFRL), in cooperation with the University of Dayton Research Institute (UDRI) and Fairchild Controls Corporation, is operating an in-house advanced vapor compression refrigeration cycle system (VCS) test rig known as ToTEMS (Two-Phase Thermal Energy Management System). This test rig is dedicated to the study and development of VCS control and operation in support of the Energy Optimized Aircraft (EOA) initiative and the Integrated Vehicle ENergy Technology (INVENT) program. This preliminary paper will discuss data collected from ToTEMS to illustrate the effect of charge level on VCS performance.

There are three aspects to refrigerant charge based on steady state operation and transient response within a subset of the operating envelope supplied with the compressor that were considered. How much this envelope expanded or contracted with refrigerant charge could not be thoroughly explored in time for this paper. The first aspect is the impact on the ability to control the desired load temperature (here, the oil outlet temperature) over the range of loads. The second is the impact on the operating parameters that affect the compressor life. The third is the affect on system COP. The following discussion expands on the three aspects.

For transients, the maximum overshoot and the time to arrive within a specified deviation of the set point are considered. Finally, the occurrence of uncontrollable operation either as oscillations that grow with time or a monotonic path to a limit point would indicate an unacceptable condition. Inherent in this is the realization that the present results are based on the configuration of ToTEMS, i.e. its component sizes and proportional-integral-derivative (PID) control settings but a hypothesis on the generalization of the results will be presented.

The effects of charge on compressor life include:

i. Not enough cooling to compressor components where the mass flow rate is too low (inlet pressure too low) or the inlet temperature is too high.

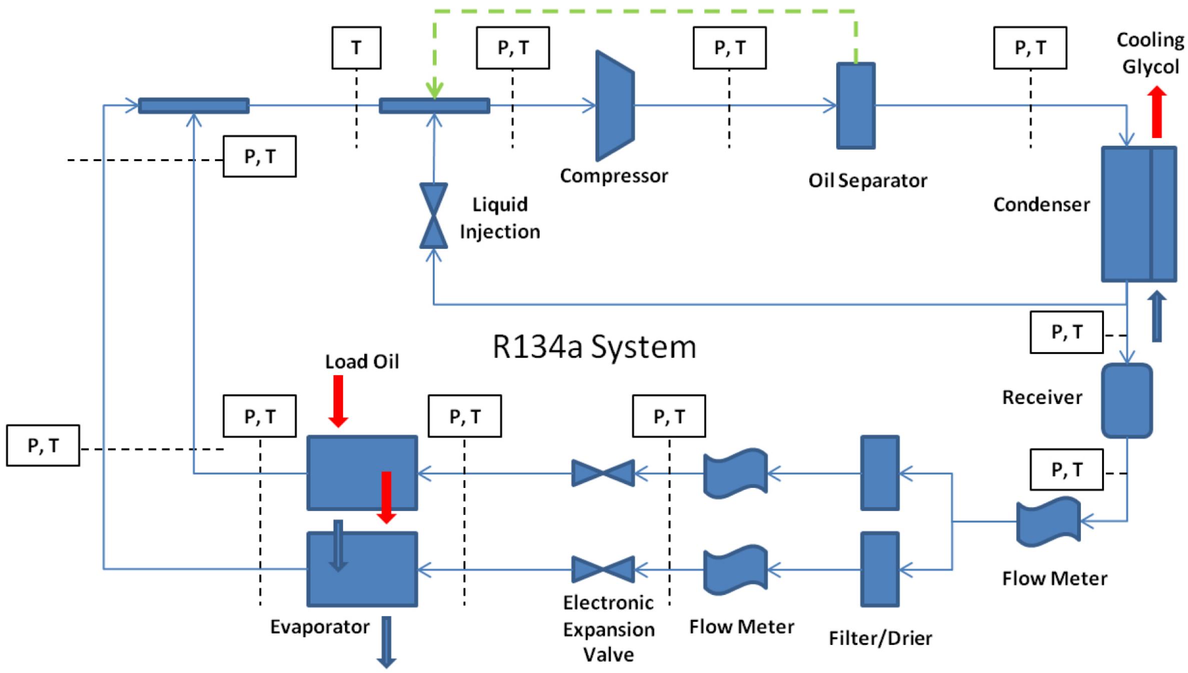

Figure 1. ToTEMS system diagram.

ii. Too high of a condenser pressure. In this case, the pressure was controlled, but in many systems it is not.

iii. Introduction of liquid into the compressor. The twin screw design has been very tolerant of small amounts of liquid entering / leaving the compressor, but other types are not.

iv. Excessive cycling of the compressor speed.

The effect of refrigerant charge on the system COP is seen in the compressor power required to remove the thermal load. Since the condenser pressure is controlled, this is linked to the subcooling (SC) and required refrigerant flow rate.

EXPERIMENT DESCRIPTION

Figure 1 shows a schematic of the ToTEMS rig used in the present work. Beginning with the variable speed screw compressor (Fairchild Controls Corporation) and moving clockwise about Figure 1, the major system components include a condenser, receiver, two electronic expansion valves, and two evaporators. The aircraft screw compressor was designed to operate with R134a. The condenser (Danfoss B3-095-72-H) has a maximum cooling capacity of 240,000 Btu/hr (~70 kW). The condenser is mounted 30° from horizontal such that the liquid refrigerant flows downward. The condenser is cooled by a 75% glycol-25% water mixture flowing (2-50 gpm at 20-150 F) from a chiller (~60 kW maximum cooling capacity). Electronic expansion valves (Emerson EX-4) regulate the R134a flow to each evaporator. Each evaporator (Fairchild Controls Corporation; rated capacity 12kW)) is heated by hydraulic oil (Mil-H-5606) which flows (1-12 gpm) from an oil heater.

Additional components include a liquid-refrigerant injection system, filter-driers, and an oil separator. Liquid injection protects the compressor from excessively high inlet temperatures and cools the compressor return oil. The filter-driers remove contaminants and water upstream from the electronic expansion valves. Oil from the oil separator is re-injected at the compressor inlet. Further details of the ToTEMS facility are given in reference [4].

In order to show the distribution of the refrigerant mass among the system components and the influence of the saturated discharge temperature (SDT), the average state of the refrigerant in each component was estimated for a saturated suction temperature (SST) of 42 F and SDT values of 115, 125, and 135 F for a SC of 36 F. The temperature at the outlet of the evaporators (56 F) was calculated as a combination of temperatures from the two evaporators with superheat (SH) values of 15F and 12F. In addition, the refrigerant in the evaporator was assumed to have a quality of 10% for simplicity. These states were used to estimate the density of the fluid using REFPROP software and, thus, the mass in each component as shown in Table 1 [5]. The condenser mass was calculated by subtracting the mass in all the other components from the total mass of 48 lb_m. The volumes were either provided by the component manufacturers or estimated from geometry.

Figure 1 shows that pressure transducers (Omega PX-409) and thermocouples are located before and after most components. Calibrated flow meters monitor the refrigerant volumetric flow rates. The thermocouples (type-T) and their immersion depth depends on the refrigerant phase. For liquid lines, the thermocouples extend to the pipe center. In the

Table 1. Component volumes and estimated mass for a 48 lb_m refrigerant charge.

Component	Vol (ft^3)	%Vol	Mass (lb_m) SDT = 115 F	Mass (lb_m) SDT = 125 F	Mass (lb_m) SDT = 135 F
LP Piping	0.278	20.8	0.291	0.291	0.291
HP Piping	0.195	14.6	13.352	13.122	12.885
Condenser	0.254	19.0	5.648	6.294	6.954
Evaporator	0.073	5.5	0.708	0.708	0.708
Receiver	0.156	11.7	11.712	11.501	11.282
Oil Separator	0.121	9.1	0.415	0.480	0.557
Filter-Driers	0.210	15.7	15.766	15.482	15.187
Compressor	0.047	3.5	0.107	0.121	0.137
total	1.334	100	48	48	48

Table 2. Instrumentation accuracy and range.

Instrument	Accuracy	Range
High Pressure-Pressure Transducers	+/- 0.76% FS	0-500 psi
Low Pressure-Pressure Transducers	+/- 0.33% FS	0-100
Glycol and Hydraulic Oil Pressure Transducers	+/- 0.16% FS	0-150
Thermocouples	+/- 0.15 F	10-250 F
Glycol Flow Meter	+/- 0.25% FS	2-50 gpm
Hydraulic Oil Flow Meter	+/- 0.11% FS	0-20 gpm
Total Refrigerant Flow Meter	+/- 1% FS	0.5-5 gpm

vapor lines near the evaporators, the thermocouples are mounted within tees that are ½″ outside the pipe diameter. The tees prevent undesirable liquid droplet impingement which would bias the measurement. Seven independent control routines are used in the ToTEMS and include condenser temperature control, evaporator SH, compressor speed, compressor inlet SH (liquid injection), and evaporator oil flow and heat load. Lastly, Table 2 lists the range and accuracy for the pressure transducers, thermocouples, and flow meters.

Experimental Procedures

The independent variables selected for study were the refrigerant mass, heater power, and SDT. In preliminary experiments, a charge of 45.5 lbm provided acceptable operation with a SDT of 125 F and, thus, was selected as the midrange value for the refrigerant mass of the system. Four masses of refrigerant were used in the experiments: 40.5, 43, 45.5, and 48 lbm. Figure 2a shows how the imposed evaporator thermal load (step and sinusoidal variation between 36 and 51%) and SDT (step changes of 115, 125, and 135 F) varied with time in the experiments. Figure 2b is an enlarged view of the period between 0 and 6000s. Each segment of the variation in thermal load had a duration of 900 s (Figure 2b). The SDT affects the liquid level and SC at the condenser which may be influenced by the system mass. The measured dependent variables are the evaporator oil inlet and outlet temperatures, DC voltage and current supplied to the compressor, refrigerant temperatures across the evaporators, and condenser exit refrigerant temperature. These variables are used to calculate COP, SC, and refrigerant SH at the evaporator exit. The refrigerant saturation temperature necessary for determination of SC and SH was estimated using REFPROP together with the measured pressure [5]. One evaporator was run with a constant oil-thermal load of 6kW while the other had a time-varying load of either 6.6, 9.3, or 12kW. The set point for the evaporator oil exit temperature was held constant. The superheat set point of one evaporator was 15 F and, the other was 12 F. Although there are two evaporators in the system, the compressor effectively treats them as a single load in the current experiments. The glycol-water mixture temperature at the condenser inlet was always 80 F.

Figure 3 represents the operating envelope for ToTEMS. Near the center of the envelope, the dashed vertical line indicates the three SDT settings that were used. The horizontal lines indicate the short term transient low pressure side SST. Figure 3 shows that the actual operating envelope is larger than the SDT and SST used in these tests. In future work, we will increase the range of the experiments.

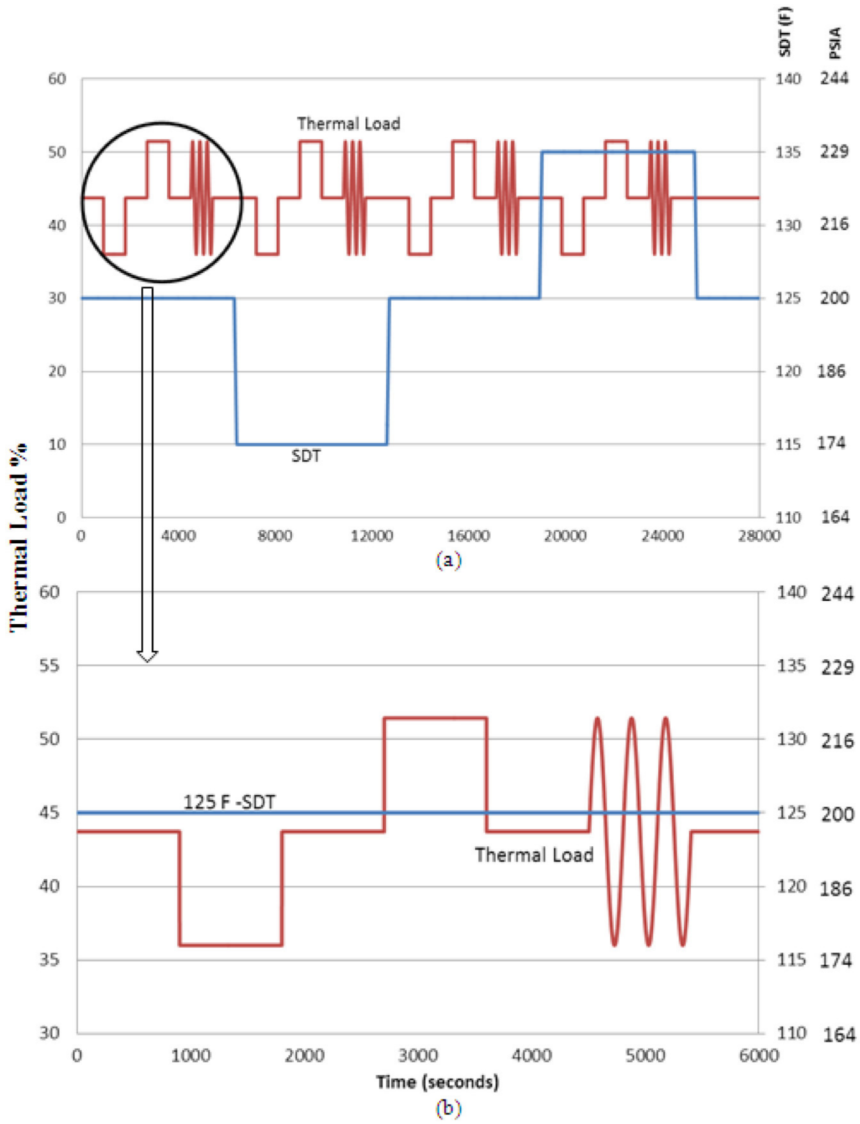

Figure 2. Testing profile for ToTEMS using four refrigerant masses (40.5, 43, 45.5, and 48 lb_m) showing (a) changes in SDT and thermal load and (b) enlarged view of the period between 0 and 6000 s.

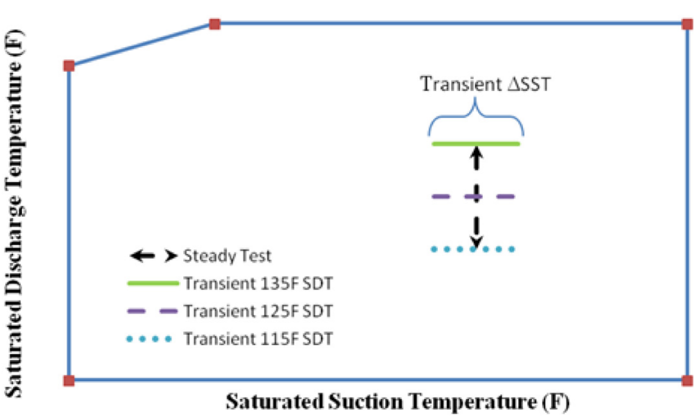

Figure 3. Representative ToTEMS compressor operating envelope. The horizontal (green and blue dashed) lines represent the SST range of values treated, and the vertical (black dashed) line is the SDT range considered.

A reference state of 125 F SDT, 42.5 F SST, 77.5% load with 15 F SH setpoint at evaporator 1, and 100% load with 12 F SH setpoint at evaporator 2 was used for testing. This reference state was repeated between all test conditions discussed above and below.

RESULTS AND DISCUSSION

Experiments were performed using the ToTEMS to assess the potential for using the SC, COP, and SH as insitu means of monitoring the refrigerant mass and to also assess what is an acceptable system mass particularly for transients. The experimental data in the following figures was recorded with all system PID controls active to maintain the desired set points.

Subcooling

Subcooling (SC = T_{sat}-T_{act} where T_{act} is measured at the condenser outlet) is typically used to ensure that the refrigerant reaching the expansion valves is entirely liquid to maintain system stability and was thought to offer potential for monitoring the refrigerant mass in ToTEMS. Figure 4 shows a plot of the steady (average) SC against the system mass obtained for different values of SDT. For a given SDT, the increase in refrigerant mass resulted in an increased residence time for the liquid refrigerant within the condenser but less available condensation area where the overall heat transfer coefficient is highest. Since the glycol flow rate increased for condensation to occur within a smaller area, the liquid refrigerant away from the condensation region will be cooler than it would otherwise be with a smaller refrigerant charge. Thus, the condenser outlet SC increased with the refrigerant charge for a given SDT as shown in Figure 4. The limit to charge addition is the eventual loss of sufficient vapor volume in the condenser which results in the loss of control of the high side pressure or SDT. The SC together with the SDT is a potential indicator of refrigerant charge for steady conditions, but this may not necessarily be true for operating conditions with large transient thermal loads.

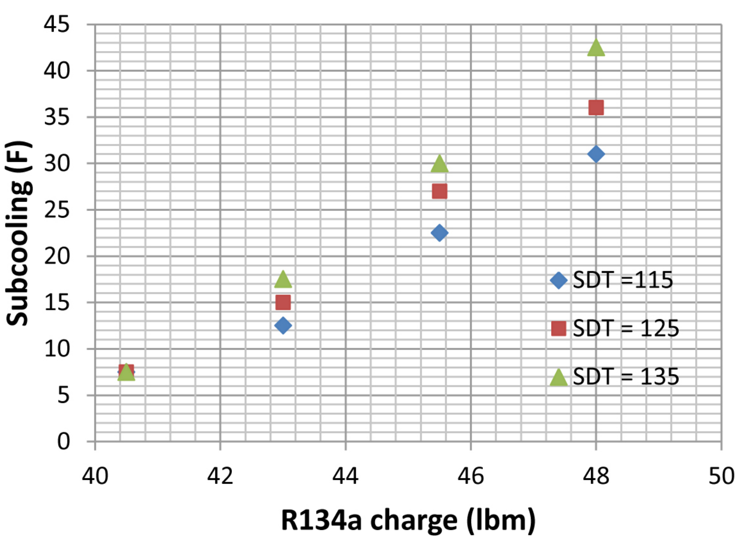

Figure 4. Subcooling as a function of refrigerant charge and SDT.

Figure 5 shows SC together with the thermal load (heater power) plotted against time for a SDT of 125 F. Near 200s, there is a step-reduction in the evaporator load from 77.5% (testing base load) to 55%. The decrease in evaporator load resulted in a relatively large reduction in SC of 5 F for a mass of 45.5 lb_m and 8 F for a mass of 43 lb_m. Smaller changes in SC occur later with other evaporator load changes for 45.5 and 43 lb_m. The SC decrease with a decrease in the evaporator load is thought to be a consequence of the control architecture. The control architecture used here is a superheat control system with total capacity control. When the load changes from 77.5% to 55%, the electronic expansion valve reduces the refrigerant flow rate to maintain the desired evaporator SH. The compressor speed also decreases as the oil temperature declines from the load reduction. This decrease in compressor speed allows the pressure on the suction side to increase and reduces the inlet flow to the condenser. This reduction in flow to the condenser diminishes the liquid refrigerant accumulation there. Consequently, there is a shorter residence time for cooling of the liquid and a decrease in the outlet condenser SC. In both the 45.5 and the 43 lbm cases, the SC begins to recover but does not have time to fully recover before the next load change occurs. In contrast for the smallest refrigerant mass, there was no significant change in SC for any variation in evaporator load. In the limit as the refrigerant mass is further decreased, the condenser becomes unable to supply sufficient liquid. For the greatest mass, the SC responds to changes in the evaporator load, but the amplitude is small. If larger refrigerant masses were used, the condenser would eventually be filled with liquid such that no cooled surface would be available for condensation.

From a controls view point, no change from the initial value of SC is desired. Thus, the lowest mass (40.5 lb_m) is preferred followed by the greatest mass (48 lb_m). Figure 5 shows that for a sufficiently large refrigerant mass, there will likely be little difference between the transient response of the minimum and maximum masses with regard to SC. Thus, the time-varying behavior of SC alone is not a useful indicator of sufficient mass. Lastly, the responses shown in Figure 5 were typical for the load reduction from 77.5 to 55% at the other SDT settings (not shown).

Coefficent of Performance

Figure 6 shows the normalized COP and evaporator load plotted against time for a SDT of 125 F. The COP curves for all masses in Figure 6 exhibit a response to evaporator load changes which contrasts the SC behavior of Figure 5. In Figure 5, the SC curves for the lowest mass and, essentially, the highest mass did not follow evaporator load curve. In addition, Figure 6 shows that the COP is greatest for the highest system mass because the higher charge enables more SC. This, in turn, reduces the mass flow rate and, consequently, lowers the compressor speed. This suggests that the mass should be further increased to maximize COP. However, the system mass is limited by the condenser becoming flooded such that the area available for condensation would be inadequate, resulting in system overpressure. Figure 6 also shows that the COP trace is noisy as it is sensitive to multiple variables such as the compressor speed and evaporator load.

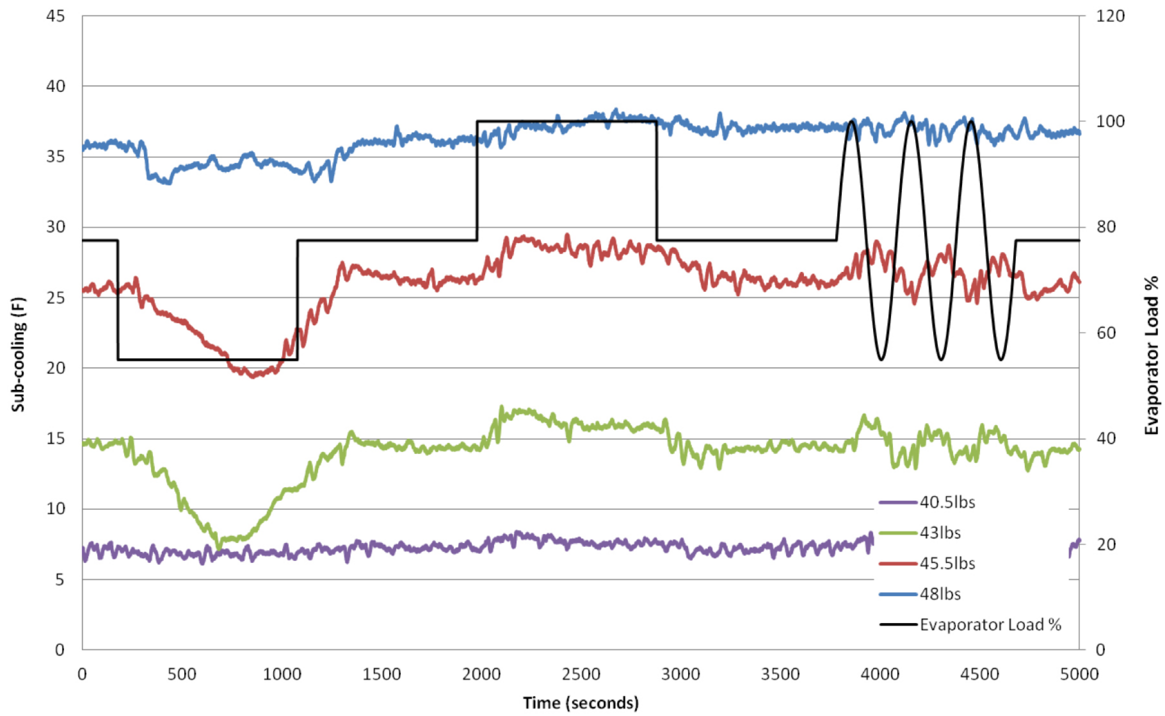

Figure 5. SC for four refrigerant masses for an SDT of 125 F and time-varying thermal load.

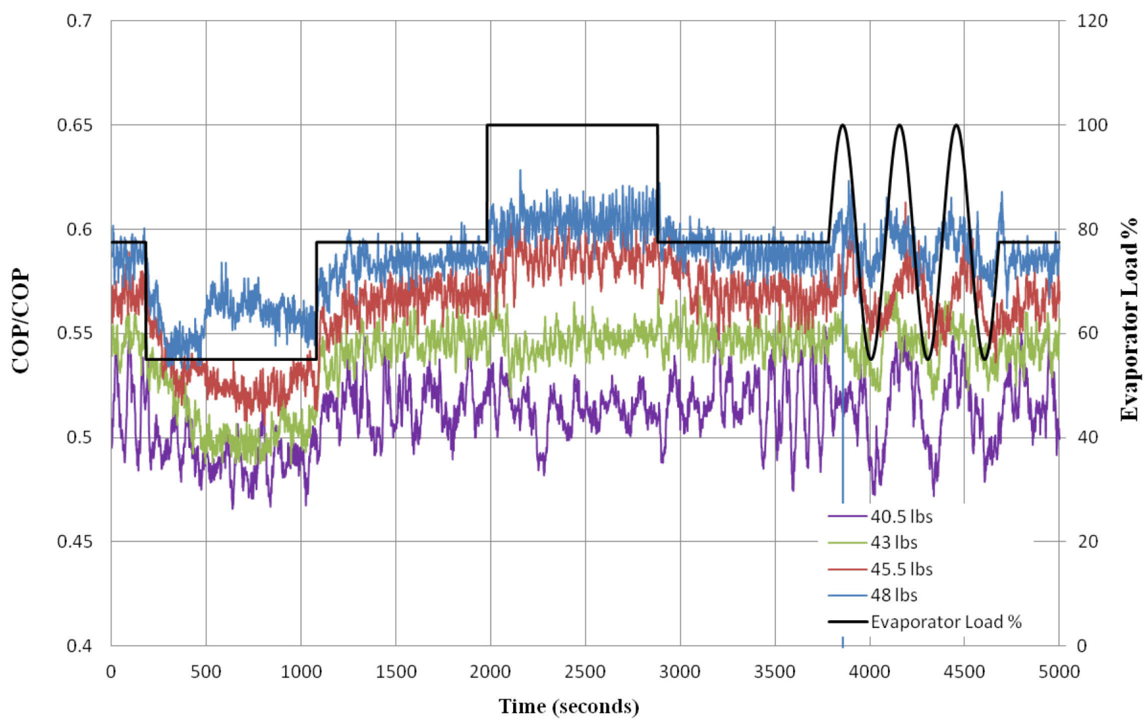

Figure 6. Normalized COP for four system masses and a time-varying thermal load for an SDT of 125 F.

The peak-to-peak variation of the COP trace for the smallest mass is large relative to those of the other masses. For example between 2000 and 3000 s, the root-mean-square (RMS) value of the COP curve for the lowest mass is 0.0102 while the other three masses have RMS values in the range 0.0070 to 0.0076. The large root mean square value is an indication of bounded behavior that is becoming difficult to control. Thus, Figure 6 shows that the large peak-to-peak variation in the COP offers a reasonable indication of inadequate refrigerant mass and has the potential to be used as an insitu indicator of refrigerant mass. Between 200 and 1100s, the evaporator load decreases as does the COP curves for all masses. In addition between 2000 and 2900 s, the COP

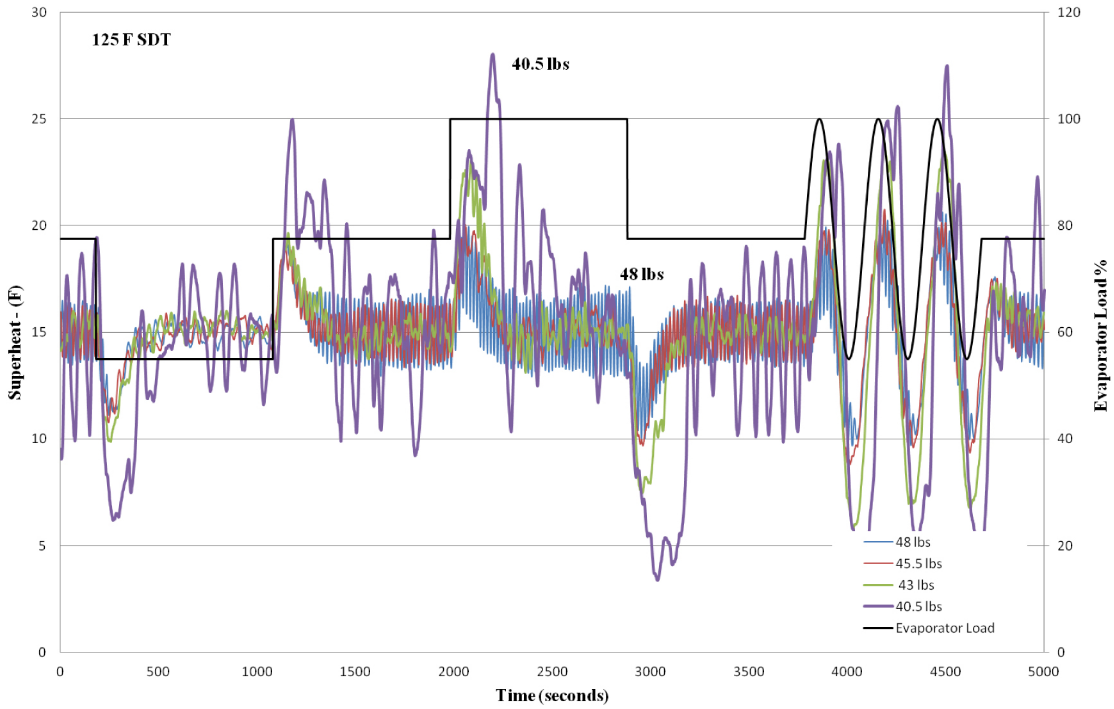

Figure 7. SH traces for four refrigerant mass charges, SDT of 125 F, and evaporator load changes.

curves for the two greatest masses increase as the evaporator load increases, but the COP curves for the two smallest masses decrease in this range. At present, we are continuing our analysis to understand this behavior.

Superheat

Figure 7 shows a time-varying SH about the setpoint value of 15 F. The 43, 45.5, and 48 lb_m masses resulted in SH excursions beyond the setpoint of about 4-5 F, with an overshoot for the 40.5 lb_m run of nearly 9 F. The recovery time of the SH by the controls to 15 F was about 200 s for all runs. Similarly, for all other evaporator load step changes in the figure, the SH controls responded, trying to minimize the amount of overshoot and recovery time. The oscillations were relatively large for the lowest mass. On the other hand, the 48lb_m system mass sometimes had larger excursions from the set point than either 43 or 45.5 lb_m (e.g., in the interval 2000-2900 s). Since it is desired to hold the excursions to as low a value as possible, it does not appear that the SH trace alone gives a clear indication of the system charge.

During the period of 2000-2900 s which coincides with a 100% evaporator load, Figure 8 shows that the expansion valve for 40.5 lb_m is being commanded to have a valve opening area that is more than twice those associated with 45.5 and 48 lb_m, and significantly greater than that for 43 lb_m. Figure 8 suggests that as the system mass decreases, the density at the expansion valves is decreased. In other words as the system mass decreases, the total phase change heat transfer is reduced within the evaporator and the expansion valve must open further to allow more refrigerant to flow through the evaporator. Thus, Figure 8 shows that there is potential to use the electronic expansion valve travel as a charge indication.

The COP, SH, and electronic expansion valve position traces all exhibit increased oscillation magnitude, and in some cases an increase in mean value for the lowest system mass. For example, the increase in the oscillation magnitude of COP had a RMS value that increased from roughly 0.0070 to 0.0102 (30% increase) with the reduction in charge. The root cause of this increase appears to be changes in the refrigerant quality supplied to the expansion valves and the evaporator. Figure 8 shows that there is a significant increase in the average opening of the expansion valve, and there is a significant increase in the amplitude of the expansion valve position oscillations. These oscillations are strictly caused by the quality of the refrigerant passing through the evaporator. The expansion valve is only attempting to control the discharge superheat.

There are a number of flow phenomena that have been observed by others in terms of the flow patterns, such as slug flow, segmented flow, etc. In this case, we did not have the ability to visually observe the flow within the evaporator. However, we can speculate with reasonable confidence that two phase flow exists at the inlet to the expansion valve with the lowest charge because: (1) there must be a proportional increase in the product of the evaporator inlet enthalpy and

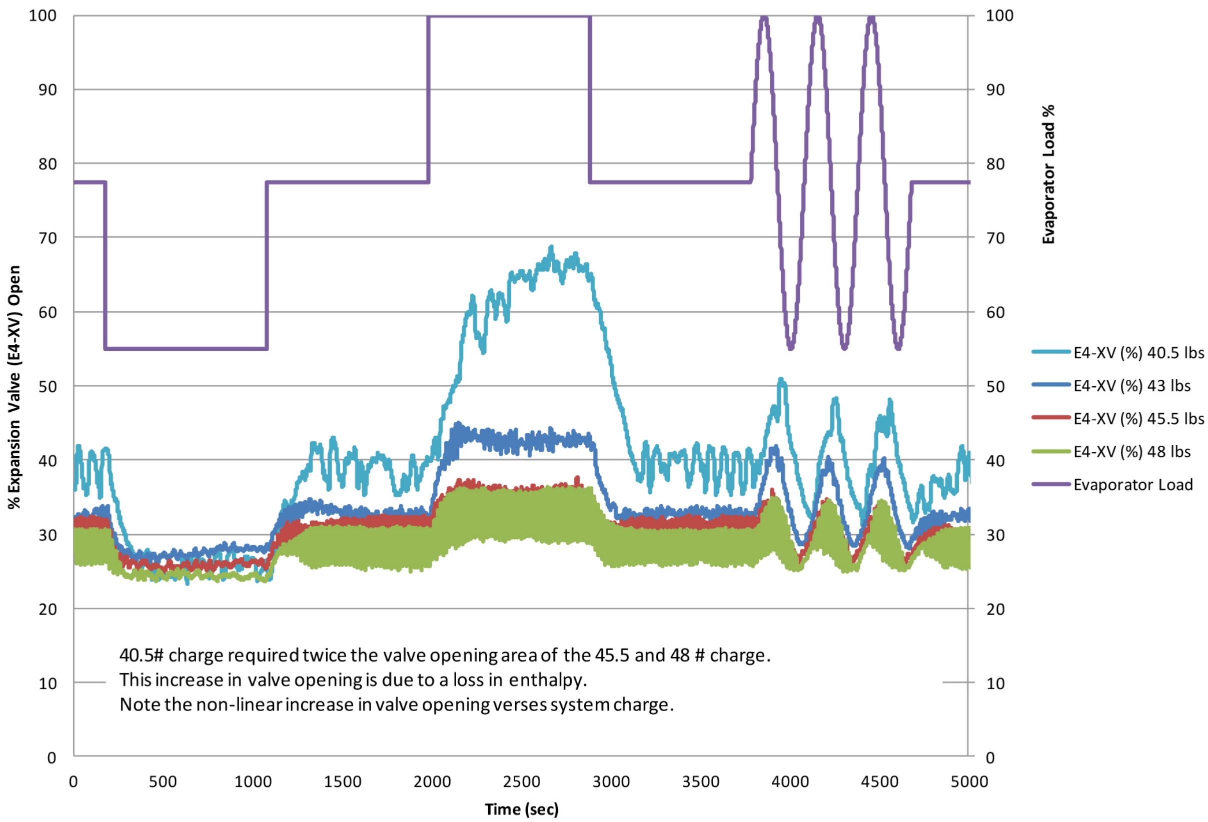

Figure 8. Percent opening of an expansion valve for different system masses and evaporator loads, SDT = 125 F.

the refrigerant mass flow rate or the mean expansion valve opening would not have increased; (2) the magnitude of the superheat oscillation and the expansion valve movement shows there is an inconsistency in the bulk quality of the fluid passing through the expansion valve. These valves are volumetric flow devices and, given that the experiment maintained a near constant pressure drop across the valve, then the only explanation is there is a change in the effective density of the fluid passing through the expansion valve.

For a fixed SDT, Table 3 shows that the enthalpy decreases about 20% as the system mass increases. However, Figure 9 shows that the mean valve position changes as much as 100% of the set point (i.e., twice the valve opening) for a system mass of 40.5 lb_m. Therefore, the average density of the fluid must be changing roughly by a factor of 2. It is very likely that the refrigerant entering the electronic expansion valve is inside the two phase region or very near the saturation line. Additional evidence that this is the appropriate conclusion is shown in Figure 9 where the mean value of the expansion valve opening decreases with the 40.5 pound charge as the SDT is increased. As the SDT increases, the pressure increases, and the expansion valve inlet enthalpy increases. Thus, further indicating that at the lower charge and lower SDT the expansion valve inlet is very near or inside the two phase region. One implication of low charge on operability could be a restriction in the minimum SDT. The ToTEMS only had difficulty controlling superheat with 40.5 pounds of charge and SDT below 135F.

Table 3. Enthalpy at the expansion valve inlet.

Enthalpy (Btu/lbm)				
	Charge in lbm			
SDT (F)	40.5	43	45.5	48
115	47.8	46.0	42.5	39.5
125	51.4	48.7	44.4	41.3
135	55.1	51.4	46.9	42.5

Figures 8 and 9 show that the change in the mean value of the expansion valve position or oscillation of the expansion valve position is a useful indicator of the system requiring additional refrigerant. Figure 10 shows that the evaporator oil discharge temperature was held within ± 2 F during all experiments. These relatively small temperature excursions show that the system was still meeting its primary requirements, but intermediate control parameters (such as SH) were only marginal.

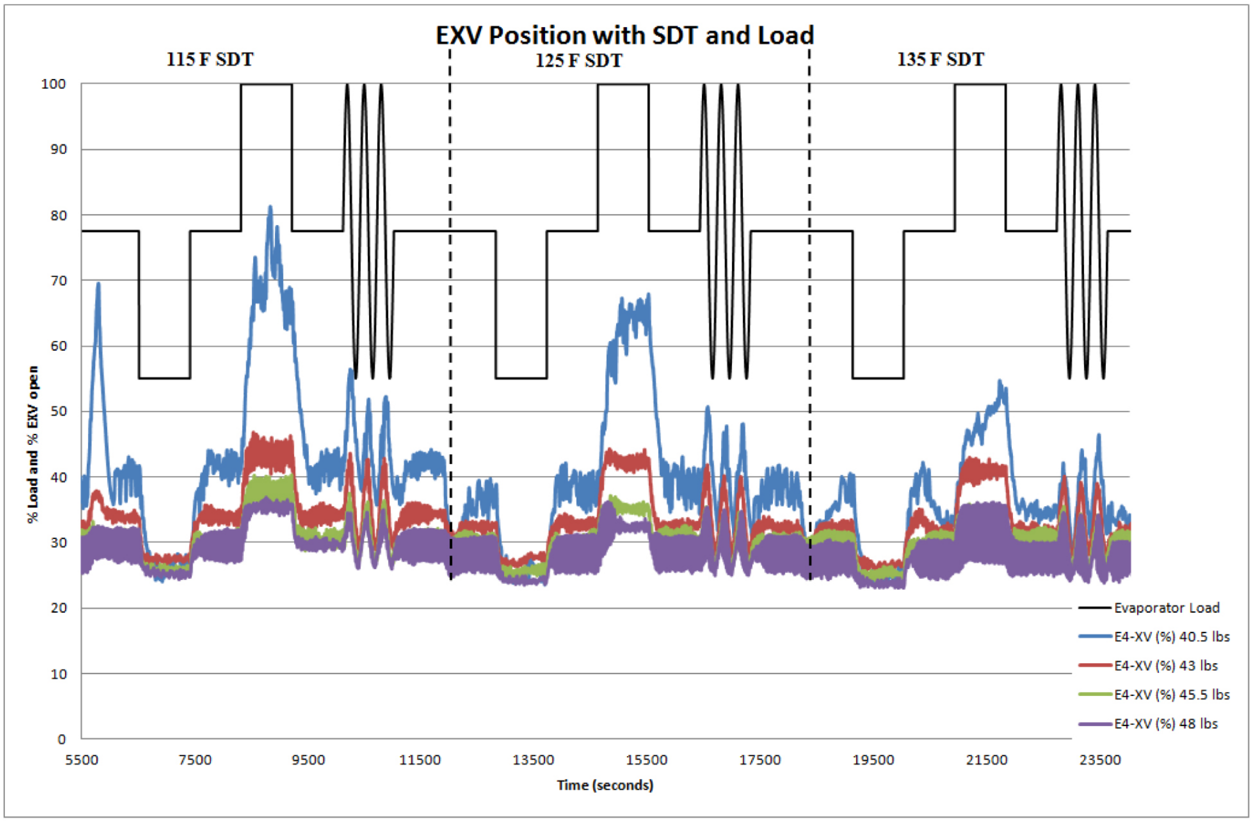

Figure 9. Thermal load and expansion valve position for different SDT values.

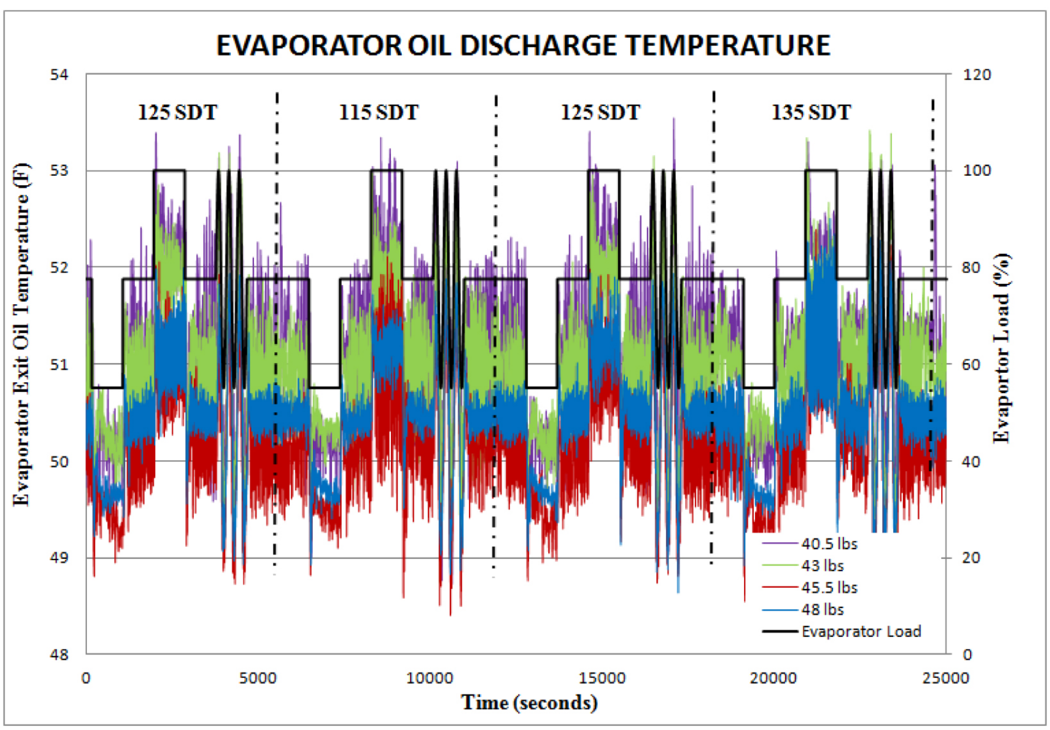

Figure. 10. Evaporator exit temperature and thermal load.

SUMMARY/CONCLUSIONS

We have presented in-situ methods of determining acceptable transient operation of the ToTEMS system. Our goal was to explore possible methods of measuring sufficient charge levels in transiently operated VCS, to better understand if there are any indicators and eventual impact of operability.

This preliminary data does shows that the RMS value of the COP, change in the mean value of the electronic expansion valve position or oscillation of the expansion valve position, or a combination of these are useful indicators of the system needing additional refrigerant. The transient SC alone was not found to be a useful diagnostic indicator. However, there is a separation among SC levels for steady state conditions that could prove to be useful for developing a correlation with charge.

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DEFINITIONS/ABBREVIATIONS

ACS - Air Cycle System

AFRL - Air Force Research Laboratory

COP - Coefficient of Performance

EOA - Energy Optimized Aircraft

CRADA - Cooperative Research and Development Agreement

INVENT - Integrated Vehicle Energy Technology

PID - Proportional-Integral-Derivative

REFPROP - NIST Transport Property Software

RMS - Root-Mean-Square

SC - Subcooling

SDT - Saturated Discharge Temperature

SH - Superheat

SST - Saturated Suction Temperature

ToTEMS - Two-Phase Thermal Energy Management System

UDRI - University of Dayton Research Institute

VCS - Vapor Cycle System

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